HYPERSONIC RESEARCH ENGINE PROJECT - PHASE IIA
FUEL SYSTEM DEVELOPMENT
FIRST INTERIM TECHNICAL DATA REPORT
ITEM NO. 55-5.01
20 MARCH THROUGH 19 JUNE 1967
NASA CONTRACT NO. NAS1-6666

AP-67-2386

30 June 1967

GARRETT

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FOREWORD

This Interim Technical Data Report is submitted to the NASA Langley Research Center by the AiResearch Manufacturing Company, Los Angeles, California, in compliance with Paragraph A, Part VII of NASA Contract No. NAS1-6666. The document was prepared in accordance with the guidelines established by Paragraph 6.3.3.2 of NASA Statement Of Work L-4947-B.

Interim Technical Data Reports are generated on a quarterly basis for the major program tasks of the Hypersonic Research Engine Project. Upon completion of a given program task, a Final Technical Data Report will be submitted.

The document in hand presents a detailed technical discussion of the Fuel System Development for the period of 20 March through 19 June 1967.

ACKNOWLEDGEMENT

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1. Gene L. Chessmore, HRE Fuel System Project Engineer, AiResearch-Los Angeles, for technical supervision and integration of report contents;

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3. John A. Boppart, Project Engineer, and Irving D. Burke, Assistant Project Engineer, both of AiResearch-Phoenix, for data on fuel valve design and development.

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1.0 INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

The Hypersonic Research Engine (HRE) is a regeneratively cooled, hydrogen fueled ramjet engine intended to be tested on the X-15A-2 aircraft. The fuel system consists of equipment aboard the X-15-2 and equipment within the HRE. Equipment aboard the X-15-2 includes liquid hydrogen fuel tanks, associated plumbing, pressurization, valves for purging and overboard dump, and other controls required for delivery of the fuel from the storage tanks to the HRE. Equipment within the HRE includes the fuel pump, pump drive, and associated plumbing and control valves to properly meter fuel flow through the fuel injectors and/or overboard fuel dump.

For development, the fuel system is separated into the following major tasks, each of which is discussed in this report:

Fuel system integration

Fuel control valves

Fuel turbopump

Aircraft fuel transfer system

The fuel system integration task includes all equipment within the HRE with the exception of the fuel control valves and turbopump. The aircraft fuel transfer task includes all fuel system equipment aboard the X-15A-2.

1.2 SUMMARY

Significant progress has been made in the areas of fuel system integration and control valve design. Problem statements and a preliminary system schematic have been prepared to define component and system integration requirements. Preliminary analyses, design concepts and design layout drawings for the fuel control valves have been completed. Fabrication and testing of breadboard and prototype components will be carried out during the next quarterly period.

The fuel turbopump and the fuel transfer system efforts have been initiated, but are presently in the preliminary concept analyses stage.

2.0 FUEL SYSTEM INTEGRATION

2.1 PROBLEM STATEMENT

To provide an integrated fuel system meeting the overall HRE requirements, it will be necessary to (1) establish the design requirements for the system components, and (2) provide for the resolution of interface requirements of the fuel system with the X-15A-2 airplane, the HRE, and the HRE controls.

2.2 BACKGROUND

As indicated in the introduction of this report, the fuel system includes the fuel transfer system aboard the aircraft and the fuel control system within the HRE. The integration requirements for the X-15A-2 portion of the fuel system apply only to the flow and pressure demands established at the X-15A-2 and HRE interface by the fuel control system within the HRE. The analytical design effort for the components in the fuel transfer system is included in the aircraft fuel transfer system effort (Section 5.0). The major portion of analytical effort under the fuel system integration task will be the establishment of design criteria for the components within the HRE.

The HRE fuel system schematic (Figure 1) shows the hydrogen fuel flow path from the aircraft interface to the combustor fuel injectors. Aboard the aircraft, the fuel flows from the liquid hydrogen storage dewars through a vacuum-jacketed transfer line to the aircraft shutoff valve and quick disconnect at the aircraft/HRE interface. From the quick disconnect, the hydrogen flows through the HRE shutoff and purge valve directly into the pump at a pump inlet pressure of 35 psia and 40°R. The hydrogen is delivered from the pump discharge at 700 psia and 49°R directly into a plenum. At this point, the flow is divided to provide proper cooling of the engine spike, inner body, and outer body.

Fuel flows through one path to the spike and inner body, where it absorbs heat to raise the hydrogen temperature to 1000°R to 1800°R, depending upon the mode of operation, and then into the fuel manifold. Fuel also flows through the outer body, where it is heated to 1000°R to 1800°R and then is directed into the turbine. A portion of this flow is bypassed around the turbine as necessary to maintain pump discharge pressure. The turbine and bypass flows are then recombined and dumped into the fuel manifold with the spike and inner body flows. The hydrogen is then directed through the fuel control valves, into the combustor injector manifolds. A dump valve and line is provided to dump excess fuel directly overboard, if the required cooling flow exceeds the engine demands.

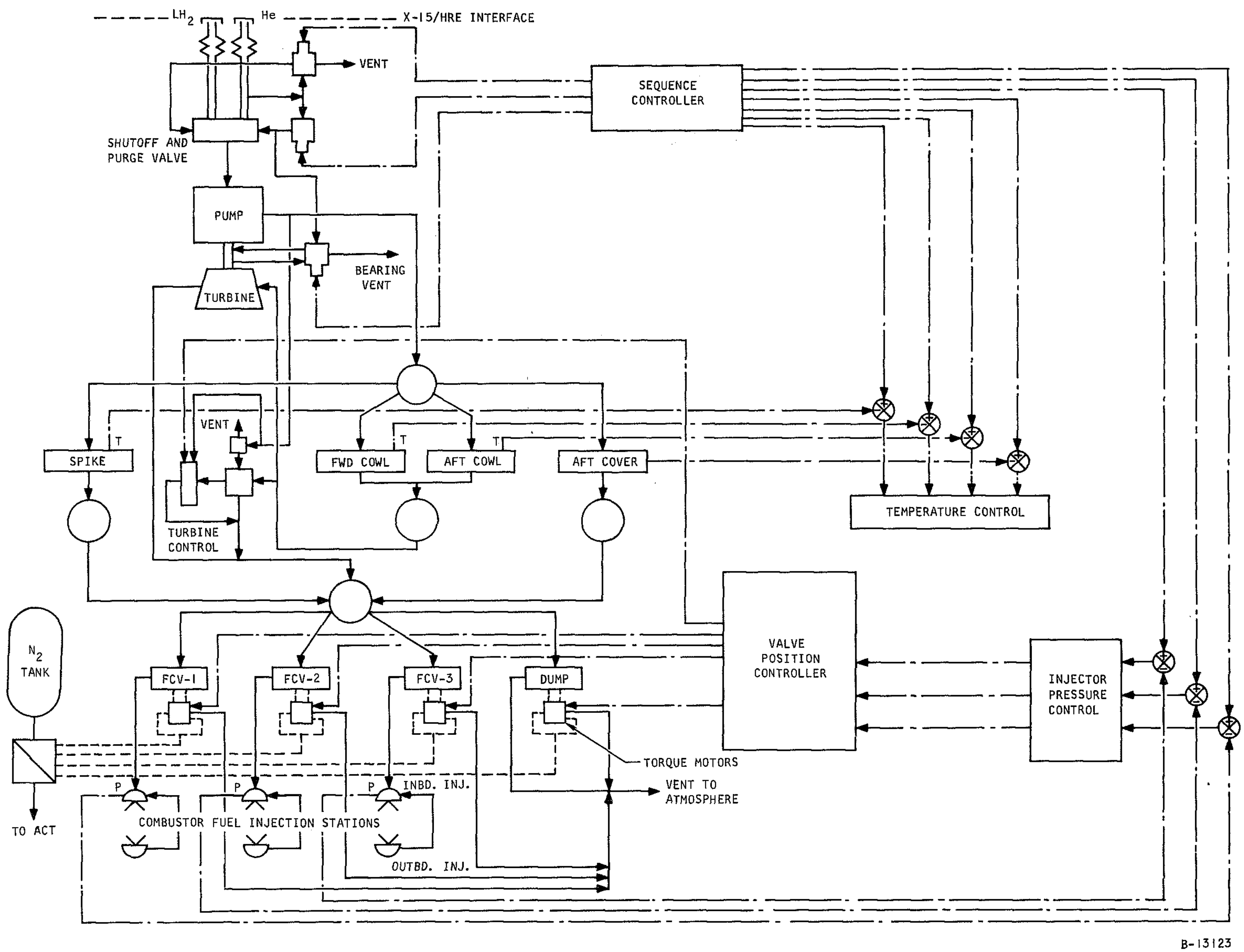

Figure 1. HRE Fuel System Schematic

2.3 OVERALL APPROACH

The approach to determine the system integration criteria is to define and perform the analytical tasks necessary to establish this criteria. These tasks include:

(1) Pressure Drop and Temperature Analysis

Establishment of an analysis program for determining pressure drop and temperature profile throughout the entire fuel flow route for various flight conditions and engine and cooling flow requirements.

(2) Fuel Control Valve Analysis

Determination of operational requirements, flow scheduling, pressure drops allowable, operating pressures and temperatures, dynamic response, etc., for valve design, including leakage requirements, flow area tolerances, and environment.

(3) Turbopump Analysis

Determination of operational requirements, operational conditions, pump and turbine flow requirements, pump and turbine power matching capabilities over the flight range, pump cooldown requirements, startup and shutdown transients, and operating environment.

(4) Purge and Valve Operating Gas Analysis

Determination of purge requirements for the storage tank and the engine, amount of purge gas available, and the conditions under which it will be available.

Determination of the amount of gas required for control valve operation and its availability.

(5) Purge and Shutoff Valve Analysis

Determination of operating requirements, flow and pressure drop requirements, operating time, and environment, for valve design.

(6) LH_2 Tankage and Feedline Analysis

Determination of general operating characteristics of tankage and feedlines, such as storage volume, pressurization gas necessary, pressure decay under flow conditions, G-effects to be considered, environmental effects to be considered, feedline pressure drop, feedline heat transfer rates, and schedule of pump inlet conditions.

(7) Failure Mode Analysis

Determination of safety and reliability requirements and preparation of a component-by-component analysis to determine system ability to meet the requirements.

(8) Analog Computer Analysis

Dynamic analysis of the fuel system (turbine control loop and fuel control loop) to help define system configuration and ability of system and components to function properly over all flight regimes.

The details of specific design problems and the analytical solutions are described in Paragraph 2.4.

2.4 ANALYTICAL DESIGN

The following is a list of symbols that are used in the calculations that follow in this paragraph:

C_P = specific heat, Btu per lb per ^{0}R

f = fuel-air ratio

f' = stoichiometric fuel-air ratio

HP_P = pump developed horsepower

HP_T = turbine horsepower

P_D = pump discharge pressure, psia

P_{IN} = valve inlet pressure, psia

PR = turbine pressure ratio

T_I = turbine inlet temperature, ^{0}R

$\dot{W}_{AIR}$ = air flowrate, lb per sec

$\dot{W}_{BP}$ = bypass flowrate, lb per sec

$\dot{W}_{H_2a}$ = fuel flowrate (actual), lb per sec

$\dot{W}_{H_2s}$ = fuel flowrate (stoichiometric), lb per sec

$\dot{W}_P$ = pump flowrate, lb per sec

$\dot{W}_T$ = turbine flowrate, lb per sec

$\dot{W}_{3456}$ = turbine control loop flowrate, lb per sec

ΔP = pressure drop, psi

γ = specific heat ratio

η_T = turbine efficiency

φ = equivalence ratio

2.4.1 Valve Flow Schedule

The development of the valve flow schedule (Table I) was based on aerodynamic and heat transfer requirements, in addition to those set forth in References 1 and 2.

The engine flows were determined from the aerodynamic data for the various altitudes and Mach numbers over which the engine must operate, and by the use of the stoichiometric fuel-air ratio

$$f' = \frac{\dot{W}_{H_2 s}}{\dot{W}_{air}}$$

where f' has a value of 0.0292 for a hydrogen-air combustion mixture. This corresponds to an equivalence ratio of 1.0.

Paragraph 4.2.2.5.1 of Reference 1 specifies that the engine shall be capable of operating at an equivalence ratio of unity, and, in addition, at equivalence ratios from 0.3 to 1.3 for subsonic combustion operation and from 0.5 to 1.5 for the supersonic combustion mode. The fuel flow for the ratio of unity and the minimum and maximum ratio for each combustion mode was obtained by:

$$\varphi = \frac{f}{f'}$$

where f is the actual fuel-air mixture ratio. Since the air flow remains the same, the above relationship reduces to merely the fuel flow ratio:

$$\varphi = \frac{\dot{W}_{H_2 a}}{\dot{W}_{H_2 s}}$$

where $\dot{W}_{H_2 a}$ is the actual fuel flow and $\dot{W}_{H_2 s}$ the stoichiometric fuel flow.

The fuel control valve flows are the flows required by the engine at the fuel injection stations.

The dump valve flow is the excess flow supplied to satisfy both the combustor and structural cooling requirements. Dump valve flow is required if the cooling φ (ratio of required cooling flow to stoichiometric flow) is greater than the combustion or engine φ (required engine flow to stoichiometric flow). The structural cooling flow was obtained by extrapolating the heat transfer design point data, and the data presented in Reference 2 for both altitude and Mach number. The flows shown for the dump valve were based on extrapolating the available data.

TABLE I

VALVE FLOW SCHEDULE

	Mach No.	Altitude--Mach No. Profile	Engine Flow, lb per sec			Valve Flow, lb per sec: FCV-1			Valve Flow, lb per sec: FCV-2			Valve Flow, lb per sec: FCV-3			Valve Flow, lb per sec: Dump Valve		
Subsonic Combustion	Engine ⊃	→	0.3	1.0	1.3				0.3	1.0	1.3				0.3		
	3.5	Max. q	0.298	0.992	1.290				0.298	0.992	1.290						
		B-B	0.258	0.858	1.115				0.258	0.858	1.115						
		*	0.125	0.418	0.544				0.125	0.418	0.544						
	4.0	Max. q	0.369	1.230	1.600				0.369	1.230	1.600						
		B-B	0.319	1.063	1.385				0.319	1.063	1.385						
		*	0.156	0.516	0.671				0.156	0.516	0.671						
	6.0	Max. q	0.338	1.125	1.462				0.338	1.125	1.462				0.320		
		B-B	0.292	0.971	1.263				0.292	0.971	1.263				0.172		
		*	0.143	0.477	0.621				0.143	0.477	0.621				---		
Supersonic Combustion	Engine ϕ	→	0.5	1.0	1.5	0.5	1.0	1.5	0.5	1.0	1.5	0.5	1.0	1.5	0.5	1.0	1.5
	6.0	Max. q	0.563	1.125	1.688	0.262	0.524	0.786	0.149	0.298	0.447	0.152	0.304	0.456	0.361		
		B-B	0.496	0.971	1.467	0.226	0.451	0.676	0.128	0.257	0.386	0.131	0.262	0.393	0.187		
		*	0.239	0.477	0.716	0.111	0.222	0.333	0.063	0.126	0.190	0.064	0.128	0.193	0.083		
	8.0	Max. q	0.439	0.879	1.317	0.439	0.879	1.317							---	0.721	0.078
		B-B	0.380	0.759	1.130	0.380	0.759	1.130							0.672	0.487	---
		*	0.189	0.379	0.569	0.189	0.379	0.569							0.337	0.352	---

*15,000 ft. above B-B

2.4.2 Valve Operating Requirements

Table 2 shows the operating requirements for the various valves used in the fuel system. The data represent the system design parameters as established in Reference 2, along with the heat transfer and engine fuel flow requirements. The fuel control valves regulating flow to the three combustor stations are designated as FCV-1, -2, -3 in this report.

2.4.2.1 Flow

The design flow for the FCV-2 valve is 1.6 lb per sec, obtained from the valve flow schedule (Table 1). Since the FCV-1 valve has a maximum flow of 1.32 lb per sec and the temperature and pressure parameters are the same as for the FCV-2, the design flow for the FCV-1 was increased to 1.6 lb per sec to provide complete commonality of hardware.

The design flow for the FCV-3 is 0.456 lb per sec, which is the maximum flow the valve is required to handle. The fuel dump valve design flow is 0.721 lb per sec, which is the flow over and above the engine fuel flow (0.879 lb per sec) under the Mach 8.0, maximum q condition. The dump valve and FCV-3 valve for which pressure and temperature requirements are identical, are being designed for maximum commonality of hardware.

The shutoff and purge valve is being designed for a flow rate of 1.70 lb per sec, as the maximum flow required per the valve schedule is 1.688 lb per sec.

The turbine bypass control valve design flow is the difference between the flow required for structural cooling of the outer body, and the flow required to power the turbine. The turbine power requirements are dictated by the pump requirements,

$$HP_T = HP_P$$

from which the turbine flow was determined by:

$$\dot{W}_T = \frac{HP_T}{1.415\ \eta_T\ C_P\ T_1 \left[1 - \left(\frac{1}{PR}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

The bypass flow was then obtained by:

$$\dot{W}_{BP} = \dot{W}_{3456} - \dot{W}_T$$

where $\dot{W}_{3456}$ is the structural cooling flow requirement in the turbine control loop. The design bypass flow is 0.5 lb per sec and the turbine inlet temperature, T_1, is 1600°R based on the heat exchanger fin design.

The maximum range of flows is based on the maximum system and engine flow requirements.

TABLE 2

VALVE OPERATING REQUIREMENTS

	FCV-1		FCV-2		FCV-3		Turbine Bypass Control		Fuel Dump		Shutoff and Purge				T/P Bearing Vent	
											Shutoff		Purge			
	Design	Range	Design	Range	Design	Range	Design	Range	Design	Range	Design	Range	Design	Range	Design	Range
Flow media	H_2		H_2		H_2		H_2		H_2		H_2		He		GH_2 GHe	
Inlet press., psia	500	400-700	500	400-700	500	400-700	575	550-700	500	400-700	50	50-60	500		500	
Inlet temp., ^0R	1600	1000-1800	1600	1000-1800	1600	1000-1800	1600	1000-1800	1600	1000-1800	40	37-42	260	260-560	100	
Flow, lb per sec	1.6	0-1.7	1.6	0-1.7	0.456	0-0.7	0.5	0-1.0	0.721	0-1.7	1.7	1-1.7	0.10			
Press. drop, psi (max)	50		50		50		150		To Ambient		15		100			
Torquemotor or Solenoid vdc	28	24-31	28	24-31	28	24-31	28	24-31	28	24-31	28	24-31	28	24-31	28	24-31
Actuation media	N_2		N_2		N_2		N_2		N_2		He		He			
Actuation Supply press., psia	600		600		600		600		600		500		500			

2.4.2.2 Pressure

The design inlet pressures of the valves are based on the pump discharge pressure of 700 psia and the line and component pressure drops:

$$P_{IN} = P_D - \Sigma\Delta P$$

The system pressure drops were based on the line size, routing, and the required cooling flows. The range of inlet pressures are based on the maximum and minimum pressures at the valves during the various flow requirements of the engine modes of operation. The system pressure drops were obtained by scaling, in accordance with the relationship: ΔP is proportional to $\dot{W}^2_{H_2}$ for flows other than the design values. The pressure drop across the fuel control valves FCV-1, -2, -3 was set at 50 psi maximum to ensure the proper fuel injector pressure and to keep the system component weight to a minimum. The turbine bypass valve ΔP was set at 150 psi maximum for proper matching with the turbine ΔP. The dump valve has the same operating parameters as the fuel control valves.

2.4.2.3 Temperature

The design fluid temperature is 1600°R based on the heat transfer requirements for structural cooling. The range of temperatures is based on the flight conditions, with a minimum of 1000°R for the lower flight operating conditions and the maximum of 1800°R imposed to protect structural integrity.

2.4.3 Pump and Available Turbine Flow

Table 3 lists the pump flows required for engine operation and structural cooling during the various modes of engine operation. The "Turbine Flow Available" column indicates the total flow that is available to the turbine, which in all cases is more than that required for turbine operation.

The available turbine flow was obtained by scaling the loop design flow calculations in accordance with the relationship:

$$\dot{W}_{3456} = \dot{W}_P \left(\frac{\dot{W}_{3456}}{\dot{W}_P}\right)_{DES}$$

for the various modes of operation.

TABLE 3

PUMP AND AVAILABLE TURBINE FLOW

Mach No.	Altitude--Mach No. Profile	Engine Flow, lb per sec			Pump Flow, lb per sec			Turbine Flow Available, lb per sec		
Engine ϕ →		0.3	1.0	1.3	0.3	1.0	1.3	0.3	1.0	1.3
3.5	(Max q)	0.298	0.992	1.290	0.298	0.992	1.290	0.1768	0.588	0.765
	(B-B)	0.258	0.858	1.115	0.258	0.858	1.115	0.1531	0.509	0.661
	(*)	0.125	0.418	0.544	0.125	0.418	0.544	0.0744	0.248	0.323
								Turb. Inlet Temp. ~ 1000°R		
4.0	(Max q)	0.369	1.230	1.600	0.369	1.230	1.600	0.2190	0.730	0.948
	(B-B)	0.319	1.063	1.385	0.319	1.063	1.385	0.1894	0.631	0.821
	(*)	0.156	0.516	0.671	0.156	0.516	0.671	0.0925	0.306	0.398
								Turb. Inlet Temp. ~ 1000°R		
6.0	(Max q)	0.338	1.125	1.462	0.658	1.125	1.462	0.413	0.667	0.868
	(B-B)	0.292	0.971	1.263	0.464	0.971	1.263	0.311	0.576	0.749
	(*)	0.143	0.477	0.621	0.143	0.477	0.621	0.114	0.310	0.368
(Subsonic) (Combustion)								Turb. Inlet Temp. ~ 1600°R		
Engine ϕ →		0.5	1.0	1.5	0.5	1.0	1.5	0.5	1.0	1.5
8.0	(Max q)	0.563	1.125	1.688	0.924	1.125	1.688	0.555	0.667	1.000
	(B-B)	0.496	0.971	1.467	0.683	0.971	1.467	0.421	0.576	0.870
	(*)	0.239	0.477	0.716	0.302	0.477	0.716	0.205	0.287	0.425
(Supersonic) (Combustion)								Turb. Inlet Temp. ~ 1600°R		
	(Max q)	0.439	0.879	1.317	1.395	1.600	1.395	0.842	0.948	0.842
	(B-B)	0.380	0.759	1.130	1.052	1.246	1.130	0.640	0.746	0.670
	(*)	0.189	0.379	0.569	0.527	0.731	0.569	0.327	0.431	0.338
								Turb. Inlet Temp. ~ 1600°R		

*15,000 ft above B-B

3.0 FUEL CONTROL VALVES

3.1 PROBLEM STATEMENT

A set of valves must be provided that are capable of regulating hydrogen flow to satisfy structural cooling and engine fuel requirements of the HRE. This set of valves includes three control valves, a dump valve, a shutoff and purge valve, a turbine bypass valve, and a turbopump bearing vent valve. Requirements for these valves are discussed below.

3.1.1 Fuel Control Valves

The fuel control valves listed below are required for the regulation of the hot hydrogen flow into the engine at three combustor injection stations and/or to the overboard dump:

Flow Control Valve, FCV-1

Flow Control Valve, FCV-2

Flow Control Valve, FCV-3

Fuel Dump Valve, FDV

These valves, represented by the schematic, Figure 2, consist of a flow control valve, valve actuator and a servo controller. During operation, a gaseous nitrogen supply pressure is applied to the servo controller inlet. The nitrogen gas passes through fixed orifices A_{1-1} and A_{2-1} and pressurizes V_1 and and V_2, respectively, while exhausting through orifices A_{1-2} and A_{2-2} past the torque motor wand and out of the servo controller.

Upon receipt of a change in electrical signal by the servo valve torque motor, the torque motor wand will seek a new position closer to either orifice A_{1-2} or A_{2-2}, (position and displacement of the wand determined by input signal) thus reducing the flow from one orifice and permitting an increased flow from the other. This change in flow will reflect in a change in pressure differential between P_2 and P_1. With the change in pressure differential, the flow control valve poppet will be displaced until a new equilibrium position is reached. At a given inlet condition to the control valve, the weight flow through the valve will be linear with the electrical input signal.

The fuel control valve is designed to provide fail-safe operation. In the event of an electrical signal failure while pressurized, the valve will open to prevent entrapment of upstream hydrogen gas.

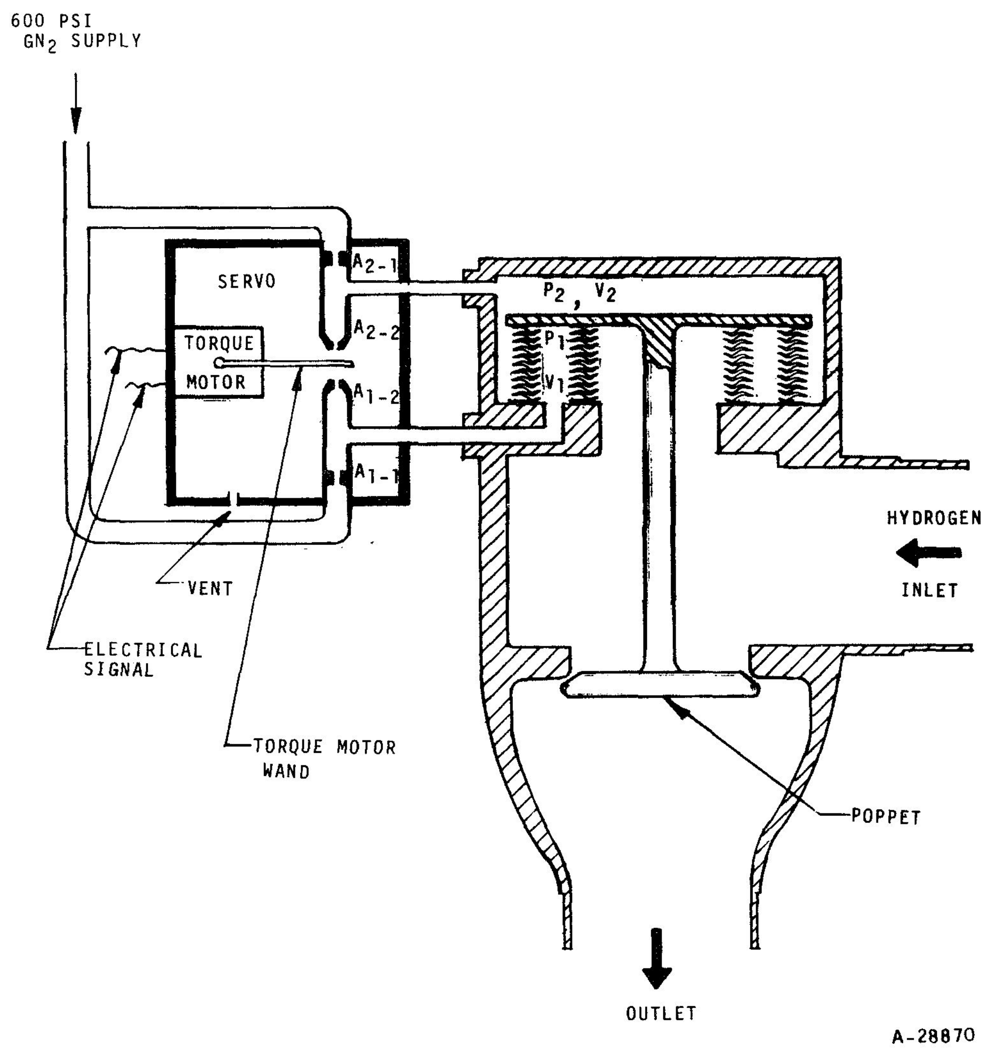

Figure 2. Fuel Control Valve Schematic

3.1.2 Shutoff and Purge Valve

A shutoff and purge valve is required to shut off the hydrogen storage and supply system in the X-15A-2 aircraft, and provide for automatic purging of the engine during the startup, shutdown, and restart modes of operation.

Figure 3 shows a schematic of the shutoff and purge valve, which consists of a hydrogen gas fuel-valving unit, with a solenoid valve-initiated, helium gas-operated actuator, and a helium gas purge valving unit with a solenoid valve-initiated, helium gas-operated actuator. Two time delay relays are employed to control the time sequencing of the two solenoid valves on starting and stopping.

Prior to engine start, the master switch is closed, which energizes time delay relay #1. The valve will allow purge flow for 3 to 6 seconds, at which time solenoid #1 will be energized. At this time purge flow will stop until the engine start switch is closed.

Upon initiation of engine start, solenoid #1 is de-energized. This opens the solenoid valve and allows the helium control gas to enter the helium purge valve actuator, thus opening the purge valve. After a period of 3 to 6 seconds, a time delay relay opens solenoid #2.

This initiates helium control gas to the hydrogen fuel valve. As the hydrogen poppet opens, the increased force supplied by spring S_2 causes the helium poppet to close, thus stopping the purge flow.

On engine shutdown, solenoid #2 is de-energized, the solenoid valve closes and vents the hydrogen valve actuator, thus stopping the flow of fuel. Simultaneously, the spring force in spring S_2 relaxes and allows the purge valve spring S_3 to open the purge valve. System purging continues for a period of 3 to 6 seconds at which time a time delay relay energizes solenoid #1, thus closing the purge valve.

Continuous purge flow may be obtained by de-energizing valve #1 by means of a separate electrical switch.

The purge and shutoff valve is designed to provide fail-safe operation. In the event of an electrical failure in the engine system, the hydrogen fuel poppet will close and the helium purge poppet will open.

3.1.3 Turbine Bypass Control Valve

A turbine bypass control valve (Figure 4) is required to modulate the amount of hot gaseous hydrogen flowing through the turbine portion of the fuel transfer turbopump in response to a sensed error signal in the pump discharge pressure.

The turbopump discharge pressure, which is always higher than the valve inlet pressure, is plumbed to the pneumatic switching element as shown, and is sensed by the switcher bellows. When the turbopump discharge pressure exceeds the prescribed value, the switcher valve begins to open. This permits flow through the actuator chamber, discharge orifice, and solenoid valve, to the

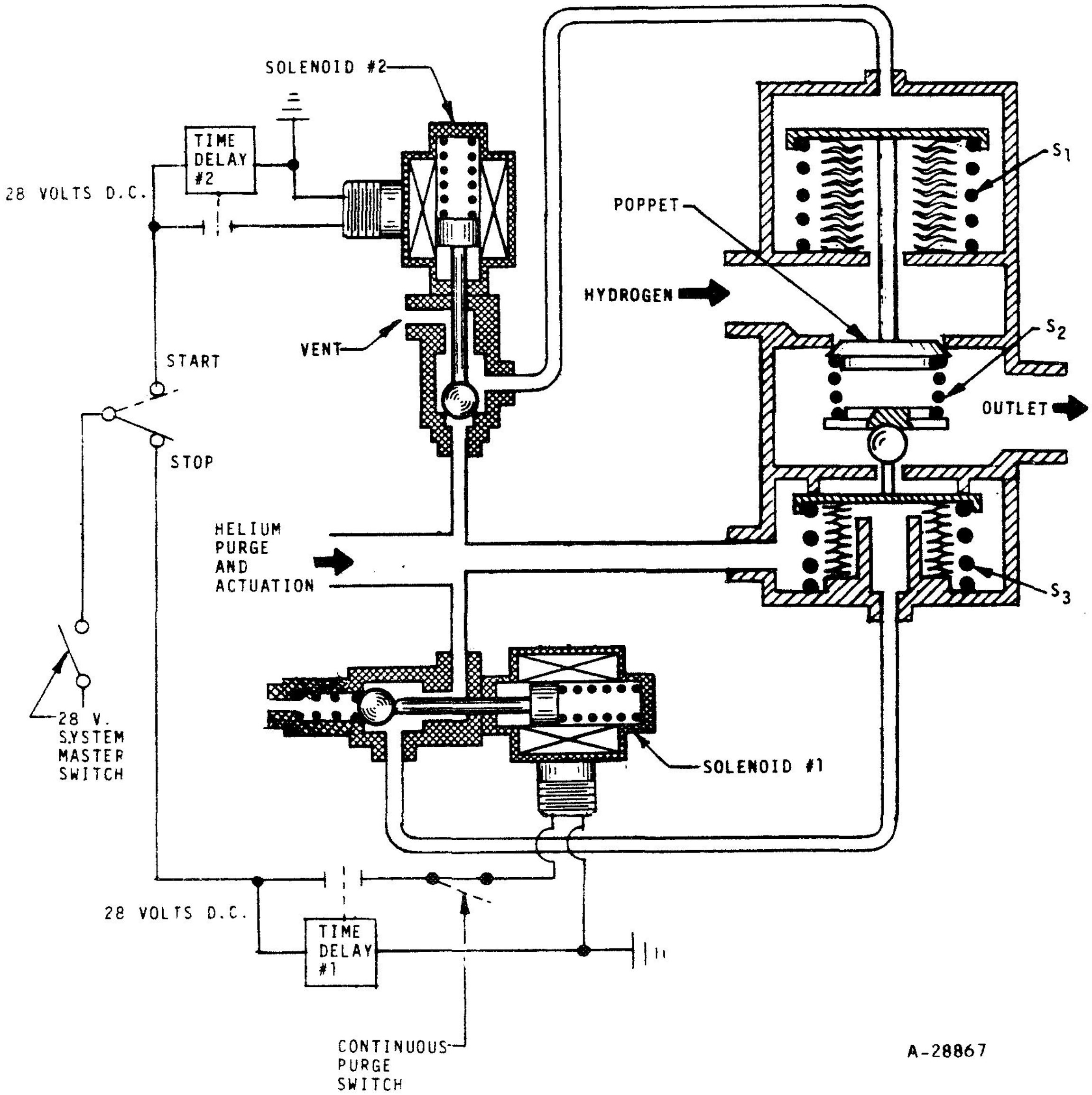

Figure 3. Shutoff and Purge Valve Schematic

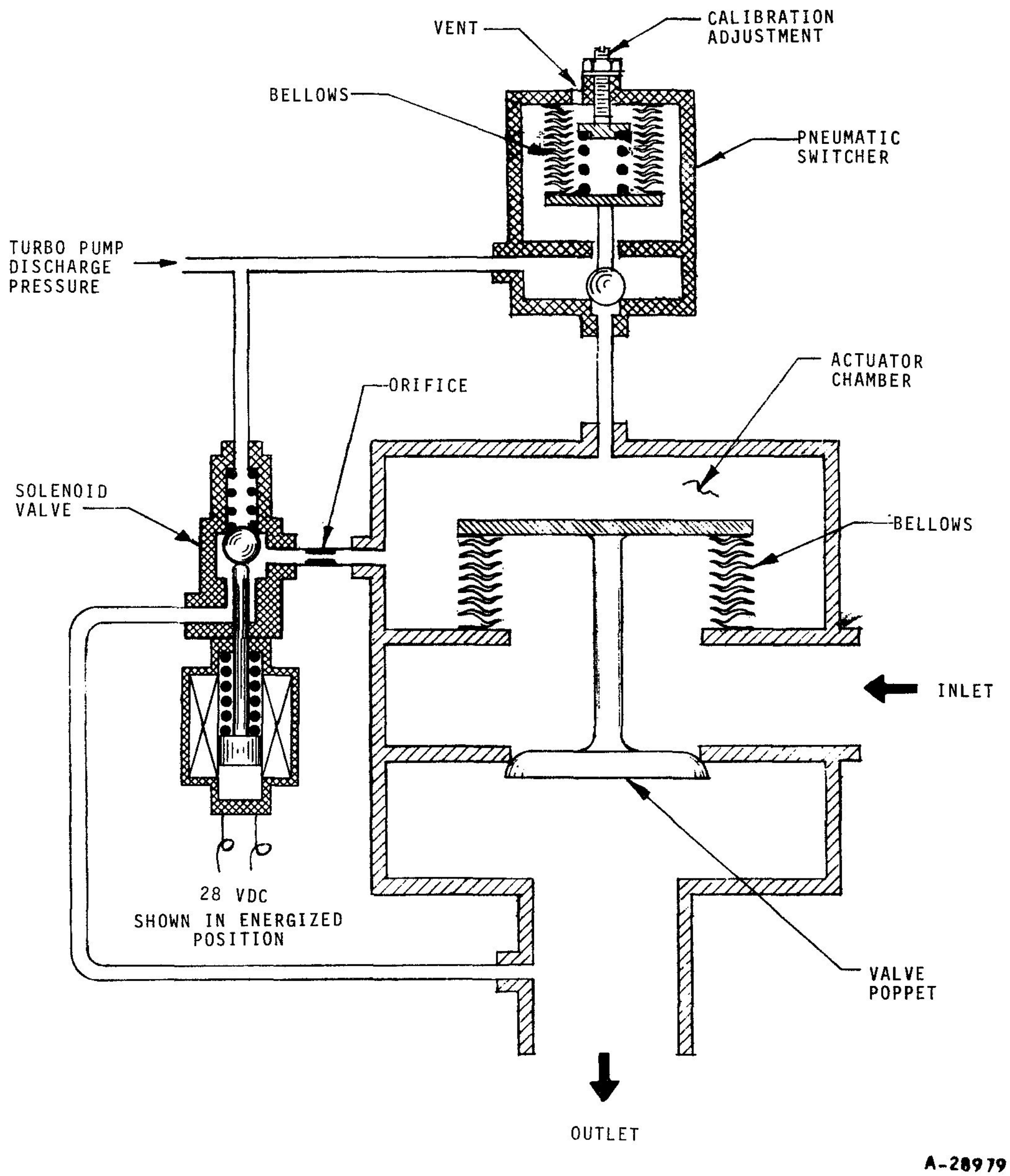

Figure 4. Turbine Bypass Control Valve Schematic

valve outlet. The resulting rise in actuator chamber pressure causes the valve poppet to modulate open. When the turbopump discharge pressure falls below the prescribed pressure limit, the pneumatic switcher closes, the actuator chamber pressure falls to the valve outlet pressure level, and the inlet pressure, acting on the bellows, forces the main valve poppet closed.

During engine operation, the solenoid valve is normally in the energized position, as shown. De-energizing the solenoid will apply full turbopump discharge pressure to the actuator chamber, causing the main poppet to open and dump the inlet pressure, thus protecting the turbopump from overspeeding in the event of electrical failure.

3.1.4 Turbopump Bearing Vent Valve

A solenoid-operated, two-position, three-way valve is required to provide purging gas to the turbopump bearing chamber when it is inoperative, and to allow venting of the chamber during turbopump operation.

3.2 BACKGROUND

The requirements for the fuel control valves are dictated by the engine performance, required structural cooling, and fuel flow regulation necessary for proper engine startup, operation, and shutdown.

Specific factors which influence the design of the various valves are required fuel flow, fuel pressure, flow accuracy, flow dynamic response, hydrogen gas temperature, structural envelope and mounting requirements, and engine environmental conditions.

The hydrogen fuel is stored as a liquid at low pressure on board the X-15A-2 vehicle. It is transferred to the HRE through vacuum-jacketed lines. At the aircraft/HRE interface, it flows through the shutoff and purge valve to the turbopump, where it is boosted to the high pressure required to provide flow through the structural cooling jackets, the engine fuel control valves, and the fuel ejectors in the engine. Both pressure and flow are determined by design characteristics of the structural cooling jackets and the fuel ejectors in order to obtain the desired engine performance.

3.3 OVERALL APPROACH

The overall approach to the valve design is to establish a basic conceptual design of the valve that will satisfy the problem statement; then to perform analytical studies to size the valve flow passages and other areas. The analytical studies are then verified by experimental testing as required.

3.4 ANALYTICAL DESIGN

3.4.1 Flow Area Sizing

The analytical design for the flow areas of the various valves is based upon the valve equation:

$$\dot{W} = \frac{CAP_1}{\sqrt{T_1}}\sqrt{\frac{2\gamma g}{(\gamma-1)R}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}$$

where $\dot{W}$ = valve flow rate, lb per sec

C = flow area coefficient

A = flow area, in.^{2}

T_1 = inlet temperature, ^0R

P_1 = upstream pressure, psia

P_2 = downstream pressure, psia

γ = specific heat ratio

g = acceleration constant

R = gas constant

The valve flow areas are controlled with spherical poppets against rounded edge seats. The valves are designed generally to provide a linear flow vs input signal relationship.

Because of the complex flow passage around the poppet, it is difficult to separate friction loss effect from that attributed to the vena contracta in determining the effective flow area, CA; therefore, experimental calibration (effective flow area vs stroke) of the poppet and seat configuration is required before the final design is complete.

3.4.2 Valve Dynamics

The valve dynamic analyses are conducted on a linearized basis about specific positions throughout the operational range. This is done to utilize the techniques of Laplace transformations and establish mathematical models for use on an analog computer.

An analysis of the FCV-1, -2 fuel control valves was conducted to aid in sizing the valve control orifices. This analysis is included as Appendix A.

3.5 DESIGN EFFORT

3.5.1 Design Concepts

The valve design effort has been concerned with establishing preliminary layouts to reflect the conceptual designs shown in Figures 2, 3, and 4.

3.5.2 Design and Fabrication Details

3.5.2.1 Shutoff and Purge Valve

Valve Body Assembly: This part will be made from a 356-T6 aluminum alloy sand casting. The casting will be machined and anodized.

Bellows Retainer Assembly: This part will be made from a 356-T6 aluminum alloy sand casting. The casting will be machined and anodized.

Bellows Assembly: Bellows assemblies consist of welded bellows sections, welded or brazed to suitable end plates. These assemblies will be fabricated from type 347 CRES. These parts will be procured from outside sources.

Valve Cover Assembly: This part will be machined from 2024-T4 aluminum alloy bar stock. The assembly will be anodized.

Springs: All springs will be made of Inconel X wire of suitable diameter. After forming, the springs will be heat treated. The springs will be procured from outside sources.

Ball: This will be a purchased part made of suitable CRES.

Solenoid Valve Assembly: The two solenoid valves will be designed and built by AiResearch Manufacturing Division of Arizona.

Items such as screws, nuts, washers, packings, and seals will be AiResearch or MS (Military Standard) parts and are to be procured from outside sources. They will be thoroughly inspected to ensure conformance to standard.

3.5.2.2 Fuel Control Valves, Turbopump Bearing Vent Valve, Turbopump Bypass Control Valve:

These valves are to be manufactured in accordance with similar processes and techniques to those described above. Design details will be submitted when preliminary layout drawings are completed.

3.6 MANUFACTURING

The manufacturing processes for the hot gas fuel control valves FCV-1, -2, FCV-3, and the turbopump bypass control valve will be based upon previous production of metal-to-metal, poppet-sealing, high temperature valves for both liquid metal and hot gas operation in which the temperature requirements are in excess of the temperature levels of this application. Fabrication of the turbopump bearing vent valve and the shutoff purge and vent valve will be based upon manufacturing concepts employed for various other cryogenic valve programs, including applications involving liquid hydrogen.

4.0 FUEL TURBOPUMP

4.1 PROBLEM STATEMENT

A fuel turbopump must be provided that will have the performance necessary for pressurization of the liquid hydrogen from storage pressure to the high pressure level necessary to obtain the required flowrates through the structural cooling paths and engine ejector nozzles.

4.2 BACKGROUND

The design of the turbopump is dictated by the engine performance and structural cooling requirements. Analysis of these requirements showed the necessity for a high pressure pumping system. Several high pressure systems besides the turbopump were considered in earlier analyses but were eliminated mainly on the basis of excessive weight. The unique feature of the turbopump is that the turboexpander drive unit uses the heat energy of the hydrogen coolant, thus eliminating the necessity for an external power source, and still allowing full utilization of the hydrogen for combustion.

4.3 OVERALL APPROACH

The overall approach to the turbopump design is to establish preliminary concepts from previous designs, to complete a detailed analytical program to size the turbine and pump units, and then to construct prototype subassembly units for experimental verification of the design parameters.

4.4 ANALYTICAL DESIGN

4.4.1 Pump

Hydrodynamic studies have been initiated to determine the preliminary pump configuration required for pumping liquid hydrogen to the HRE. The maximum pump performance requirements are as follows:

Inlet total pressure, P_1	35.0 psia
Inlet total temperature, T_1	40^oR
Outlet total pressure, P_{ex}	700 psia
Mass flow, $\dot{M}$	1.70 lb per sec

Under these design conditions the pump pressure rise will be 665 psia, or a pressure ratio of approximately 20 to 1. The liquid hydrogen vapor pressure, P_v, at the specified inlet conditions, is 26.0 psia and the net positive suction head, H_{SV}, is 9.0 psia (308.5 ft of liquid hydrogen, based on a liquid density of 4.20 lb per cu ft).

Using the pump pressure head of 665 psia (22,800 ft head of LH_2), and an inlet volume flow of 182 gpm, the pump specific speed, N_S, can be calculated:

$$N_s = \frac{N\sqrt{Q}}{\Delta H^{\frac{3}{4}}} = \frac{N\sqrt{182}}{(22{,}800)^{\frac{3}{4}}} = 0.007277N$$

where ΔH = Pressure head, ft of LH_2

N = Pump speed, rpm

Q = Volume flow, gpm

In order to calculate the pressure head; an impeller tip speed of 1300 ft per sec was assumed. The value of the pressure coefficient, ψ, for this tip speed is 0.435 and N_S, for optimum conditions, is approximately 2500. This value of N_S is high for a low mass-flow, high head-rise pump, giving a rotor speed of 344,000 rpm. When the rpm is restricted to 75,000, N_S drops to 545 for which Figure 5, an empirical curve from the AiResearch Pump Design Handbook, yields a value of 0.60 for the pressure coefficient, ψ. The corresponding head rise capability of 938 psia considerably exceeds the required 675 psia.

At 1135 ft per sec tip speed, which corresponds to a pressure coefficient, ψ, of 0.57, the pump diameter rpm variation is given below:

Rpm	Diameter (in.)
80,000	3.25
75,000	3.47
50,000	5.20
20,000	13.00

For 75,000 rpm, the diameter is 3.47 in.; thus for these conditions a 3.2 to 5.2 in. pump diameter appears feasible, depending upon the stress-and bearing-dictated rpm limits. The Reynolds number is greater than 10^7, thus at a specific speed of 545, an efficiency of approximately 0.76 is empirically indicated for a large impeller. (See Figure 6, reproduced from the AiResearch Pump Design Handbook.)

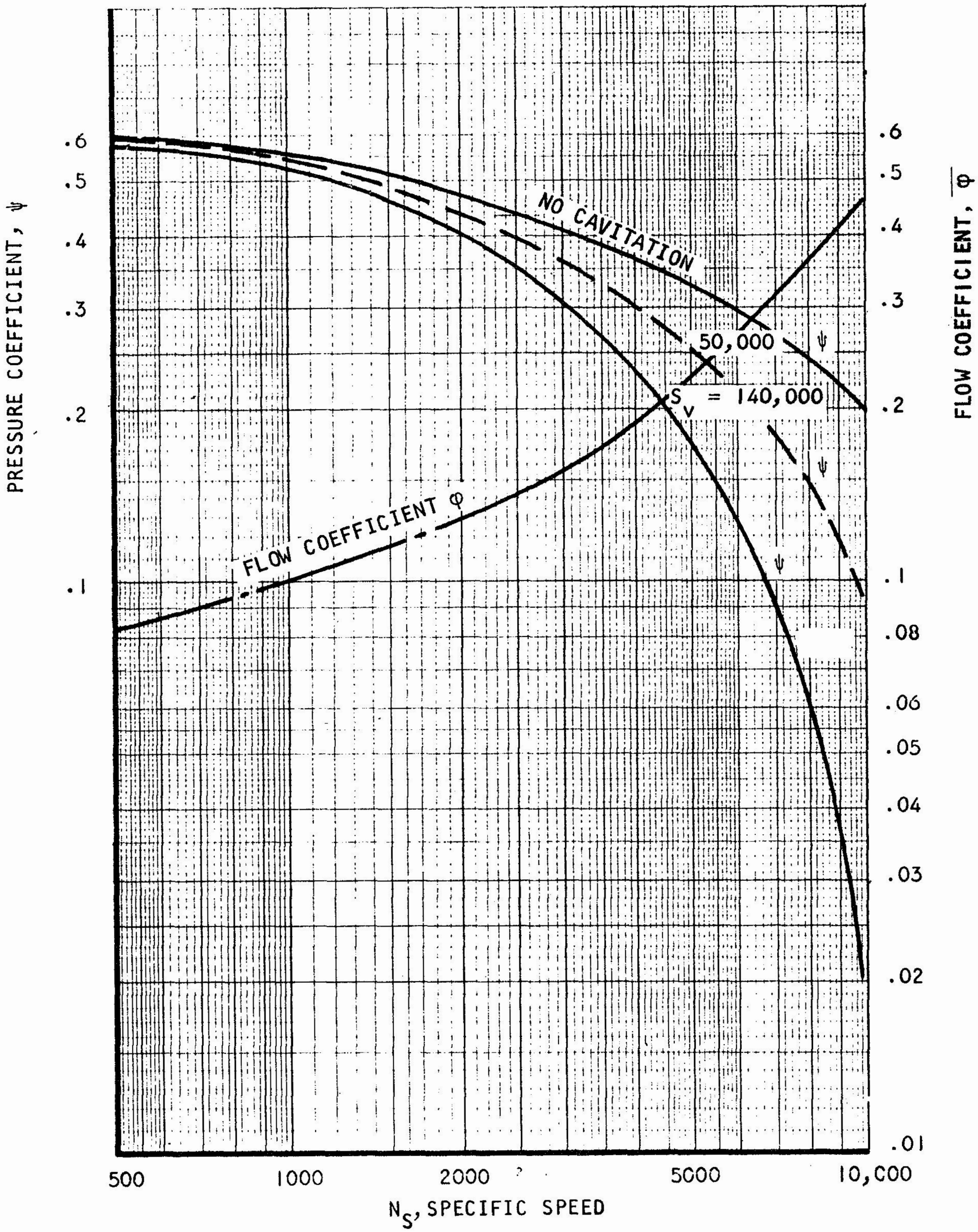

Figure 5. Optimum Performance Values of φ and ψ

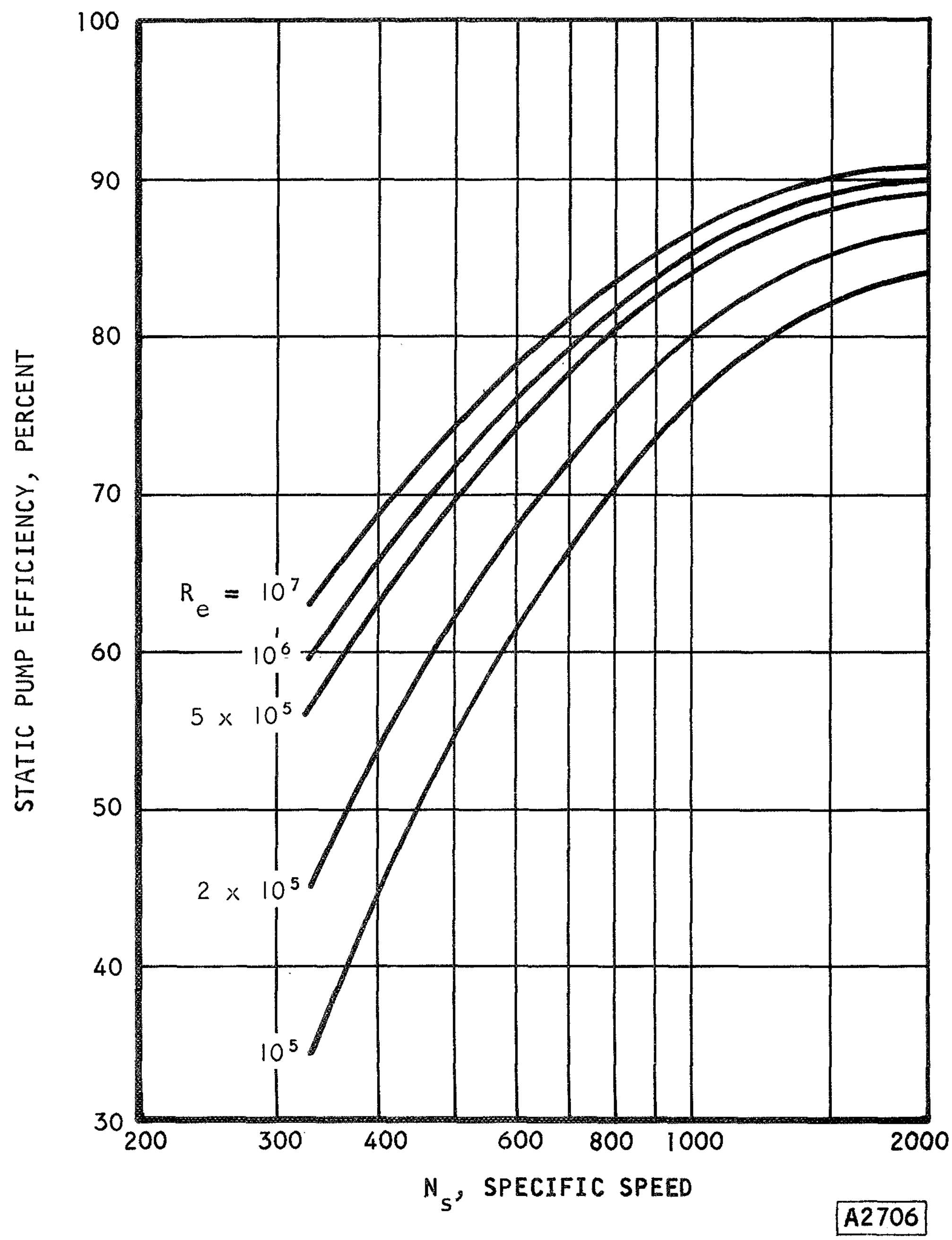

Figure 6. Static Pump Efficiency vs Specific Speed at High Reynolds Numbers for Centrifugal Pumps

The pump horsepower is calculated by:

$$H.P. = \frac{\Delta H \times Q}{\eta \times 1714} = \frac{665 \times 182}{0.76 \times 1714} = 92.8.$$

where ΔH = Pressure head, psia

Q = Volume flow, gpm

η = Efficiency

Thus, the pump power requirement is approximately 100 HP, allowing for a somewhat lower pump efficiency.

In order to determine the amount of cavitation to be expected in this pump design, it is first necessary to calculate the pump suction specific speed, S, given by the equation:

$$S = \frac{N\sqrt{Q}}{(H_{sv})^{\frac{3}{4}}}$$

where N = Pump speed, rpm

Q = Volume flow, gpm

$H_{sv} = P_1 - P_v$ = Net inlet head, ft of LH_2

In this pump design, H_{sv} = 308.5 (9 psia), then:

$$S = \frac{75{,}000\sqrt{182}}{(308.5)^{\frac{3}{4}}} = 13{,}750$$

The relatively low value of suction specific speed required indicates that pump cavitation will not present any unusual design problems.

A satisfactory value of the Thoma (cavitation) parameter, σ, is also obtained by:

$$\sigma = \frac{H_{sv}}{\Delta H} = \frac{308.5}{22{,}800} = 0.0135.$$

Thus, a feasible pump design can now be defined by the following parameters:

Impeller Dia = 3.47 in.	ΔP = 665 psia
N = 75,000 rpm	HP $\simeq$ 100
$\dot{M}$ = 1.70 lb per sec	N_s = 545
η = 0.70 to 0.75	S = 13,750
U_T = 1135 ft per sec	

In order to form a preliminary estimate of pump capability within package limits, it is possible to show a variation of size (dia) with speed and efficiency. Pump efficiency as a function of specific speed is taken from Figure 6. The following table summarizes the range of efficiencies and impeller diameters for two values of tip speed and a range of specific speeds.

TABLE 4

PUMP EFFICIENCIES FOR VARIOUS SPECIFIC SPEEDS

N_s, Specific Speed	N, rpm	ψ, Pressure Coefficient	η, Efficiency	U_t, Tip Speed, fps	Impeller Diameter, in.
300	41,222	0.607	0.60	1100	6.10
400	54,963	0.607	0.68	1100	4.59
500	68,704	0.607	0.74	1100	3.66
600	82,444	0.607	0.78	1100	3.06
400	54,963	0.375	0.68	1400	5.84
500	68,704	0.375	0.74	1400	4.66
600	82,444	0.375	0.78	1400	3.89

These data are used to plot the variation of diameter with rotational speed, tip speed, and efficiency shown in Figure 7 for a ΔP of 665 psia, a mass flow of 1.70 lb per sec, and required HP of approximately 100, depending on efficiency.

The values of specific speed are low because of the low mass flow resulting in a low flow coefficient and a high pressure coefficient. The rotor tip speed, which may be limited by drive turbine requirements, is assumed to be 1100 to 1400 ft per sec in the present case. This results in a range of efficiencies from 0.60 to 0.74. These values are uncorrected for size, which will affect the efficiency level at diameters below 4.0 inches. For tip speeds of 1100 to 1400 ft per sec, it is feasible to design a pump from approximately 3.5 to 6.5 inches in diameter, depending on the desired size and design performance limits.

4.4.2 Turbine

Design conditions for a turbine that will be compatible with the previously designed pump are as follows:

Fluid:	Gaseous hydrogen
Inlet pressure:	575 psia
Inlet temperature:	$1600^{\circ}R$

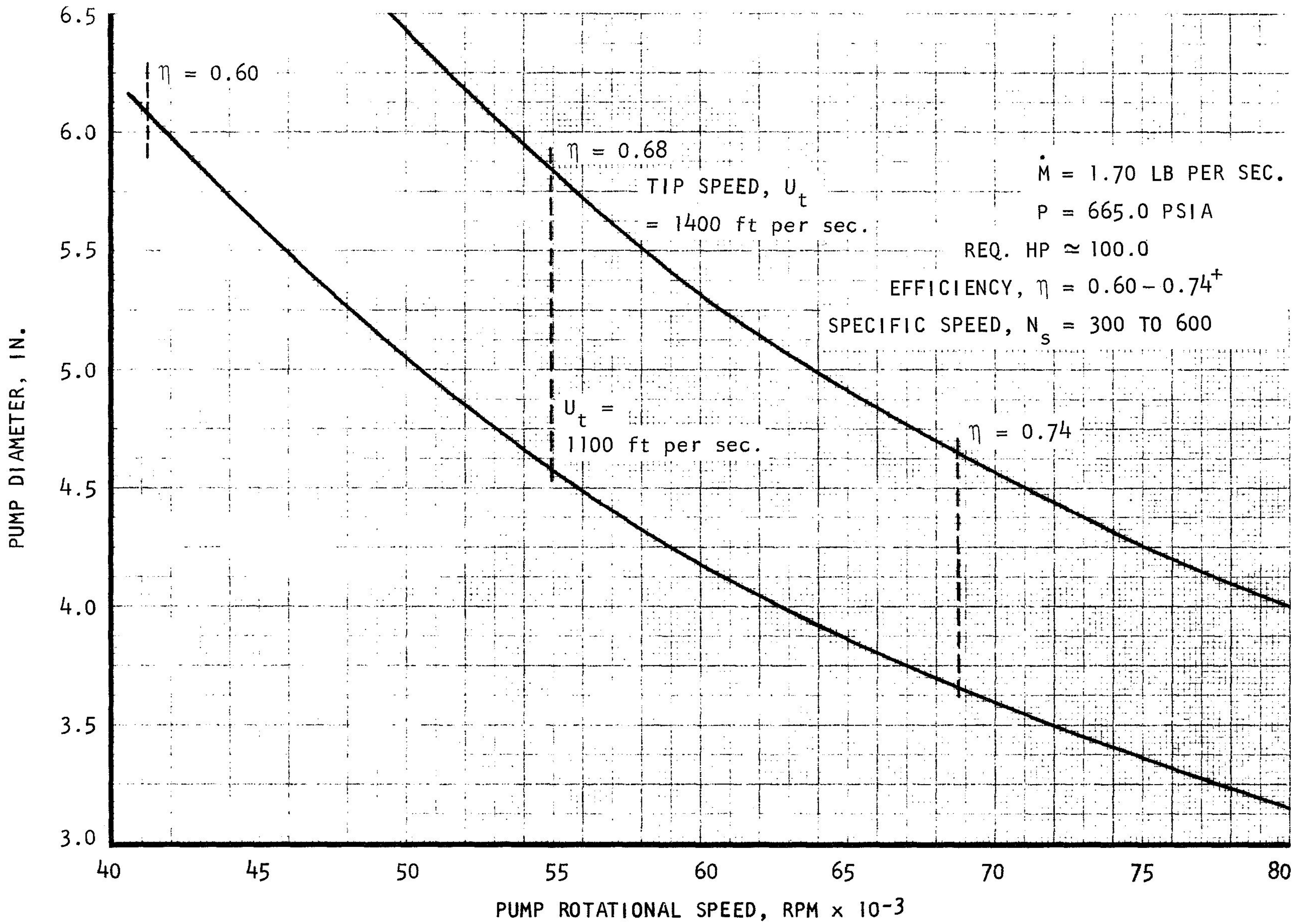

Figure 7. Preliminary HRE Liquid Hydrogen Pump Performance

Flow rate: 1.0 lb per sec

Pressure drop: 150 psi (maximum)

At these design conditions, the minimum required component efficiencies are relatively low, as shown by:

$$\eta_P \cdot \eta_T \geq 0.12,$$

assuming a maximum available turbine pressure drop of 150 psia and a mechanical efficiency of 0.90.

If the calculated pump efficiency of 0.70 to 0.75 is assumed, the minimum required turbine efficiency is 0.172. A wide range of turbine designs are capable of meeting this minimum requirement.

At the design point, the hydraulic performance of both the pump and turbine increase with rotational speeds up to approximately 90,000 rpm; however, in order to minimize development problems with seals and bearings, the speed will be limited to 80,000 rpm maximum.

The final choice of turbine type and design speed will not be determined until after careful consideration of all the requirements and design criteria involved.

The turbine can be an axial or radial type, depending upon thrust, weight, size, cost and reliability considerations.

The design speed will be dependent upon several factors, including the wheel diameter and peripheral velocity. Using a minimum turbine pressure drop, (maximum turbine efficiency) requires a peripheral velocity of approximately 1500 ft per sec, and assuming this quantity as the design peripheral velocity, the turbine wheel diameter will vary inversely with the rotational speed:

N, rpm	Diameter, in.
40,000	8.60
50,000	6.86
60,000	5.72
70,000	4.90
80,000	4.29

The minimum possible turbine pressure drop is approximately 35 psi, corresponding to a turbine efficiency of 0.85, a speed of approximately 80,000 rpm, and a peripheral speed of 1500 ft per sec.

4.5 SUMMARY OF ANALYTICAL EFFORT

The turbopump design speed finally selected will probably be in the range of 50,000 to 80,000 rpm. The pump impeller will have a diameter of 3 in. to 5 in. and will have a peripheral speed of approximately 1135 ft per sec. The turbine will have a diameter of approximately 4 in. to 7 in. and a peripheral speed of 1500 ft per sec or less, depending upon the importance of minimizing the turbine pressure drop.

At this time, no serious design problems exist, unless size and weight limitations later require the design to utilize excessively high rotational speeds.

Various off-design conditions will be investigated in order to determine the turbine pressure drop, turbopump rotational speed, and the type of unit that can meet all off-design points as well as performance requirements.

5.0 AIRCRAFT FUEL TRANSFER SYSTEM

5.1 PROBLEM STATEMENT

A liquid hydrogen storage system and helium pressurization and purge system is required in the X-15A-2 airplane for operating the HRE. The design effort will cover the system, but not the detail design of system components. The design analysis will be supported by tests utilizing substitute fluids.

An air vehicle hazard and safety analysis will be conducted, consistent with current flight safety requirements for the X-15A-2 airplane, which will establish the personnel maintenance operations and the operational and design requirements necessary for the safety of the X-15A-2 and the HRE.

Upon completion of the above design analyses, a definitive test specification will be prepared for a test program to determine the adequacy and compatibility of existing X-15A-2 fuel system components. In addition, instrumentation requirements will be defined so that all instrumentation necessary to implement this test program can be provided.

A reliability analysis of the liquid hydrogen storage, transfer, pressurization, and purge systems will be conducted. This analysis will be based on the design analyses and supporting test efforts outlined above.

5.2 STATUS SUMMARY

North American Aviation Incorporated was selected as subcontractor for the engineering effort relating to the fuel transfer and storage systems because of their technical familiarity with the X-15A-2 aircraft and their existing functional organization. Accordingly, on 10 May 1967, the subcontract was let and activity started. Effort in the report period with respect to the fuel system has been limited to review of the engine requirements described herein.

REFERENCES

1. L-4947-B NASA Statement of Work--Hypersonic Research Engine Project for Phase IIA; Development, Design Construction and Testing of the Hypersonic Ramjet--Langley Research Center, Unclassified, dated 15 January 1967.

2. AP-66-0168-2 Hypersonic Ramjet Experiment Project, Phase I, Appendix A, Preliminary Design Report (U) Volume VI, CONFIDENTIAL.

APPENDIX A

DYNAMIC ANALYSIS FOR FLOW CONTROL VALVE FCV-2

Dynamic analysis for the hydrogen flow control valve was conducted on a linearized basis at specific positions throughout the operational range. Dynamic characteristics were investigated with and without valve downstream effects. The system under consideration is shown schematically in Figure A-1. This system may be represented in block diagram form as shown in Figure A-2.

The valve response, obtained directly from Figure A-2, assuming negligible downstream dynamic effects, is given by:

$$\frac{x}{i} = \frac{K\,(\tau_3' S+1)}{(\tau_1' S+1)(\tau_2' S+1)\left(\frac{S^2}{\omega_n'^2} + \xi\,\frac{S}{\omega_n'} + 1\right)} \qquad (1)$$

where x = valve displacement, in.

i = torque motor current, amp

K_2 = servo gain, lb per in.

k = equivalent valve spring rate, lb per in.

K_{TM} = torque motor gain, in. per amp

K = $K_{TM}K_2/k$, in. per amp

τ' = time constant, sec

ω_n' = natural frequency, rad per sec

ξ = damping ratio

S = Laplace operator, sec^{-1}

The time constants, natural frequency, and damping ratio shown in equation (1) are functions of valve position and servo inbleed orifice diameter d_{OI}. The dynamic response of equation (1) is characterized primarily by the time constants τ_1', τ_2', and τ_3', as their reciprocals are several orders of magnitude less than the second order natural frequency ω_n'. These time constants attain their largest values (lowest break frequencies) when the valve is fully open. Tabulation of the above parameters as a function of inbleed orifice diameter d_{OI} is given in Table A-1.

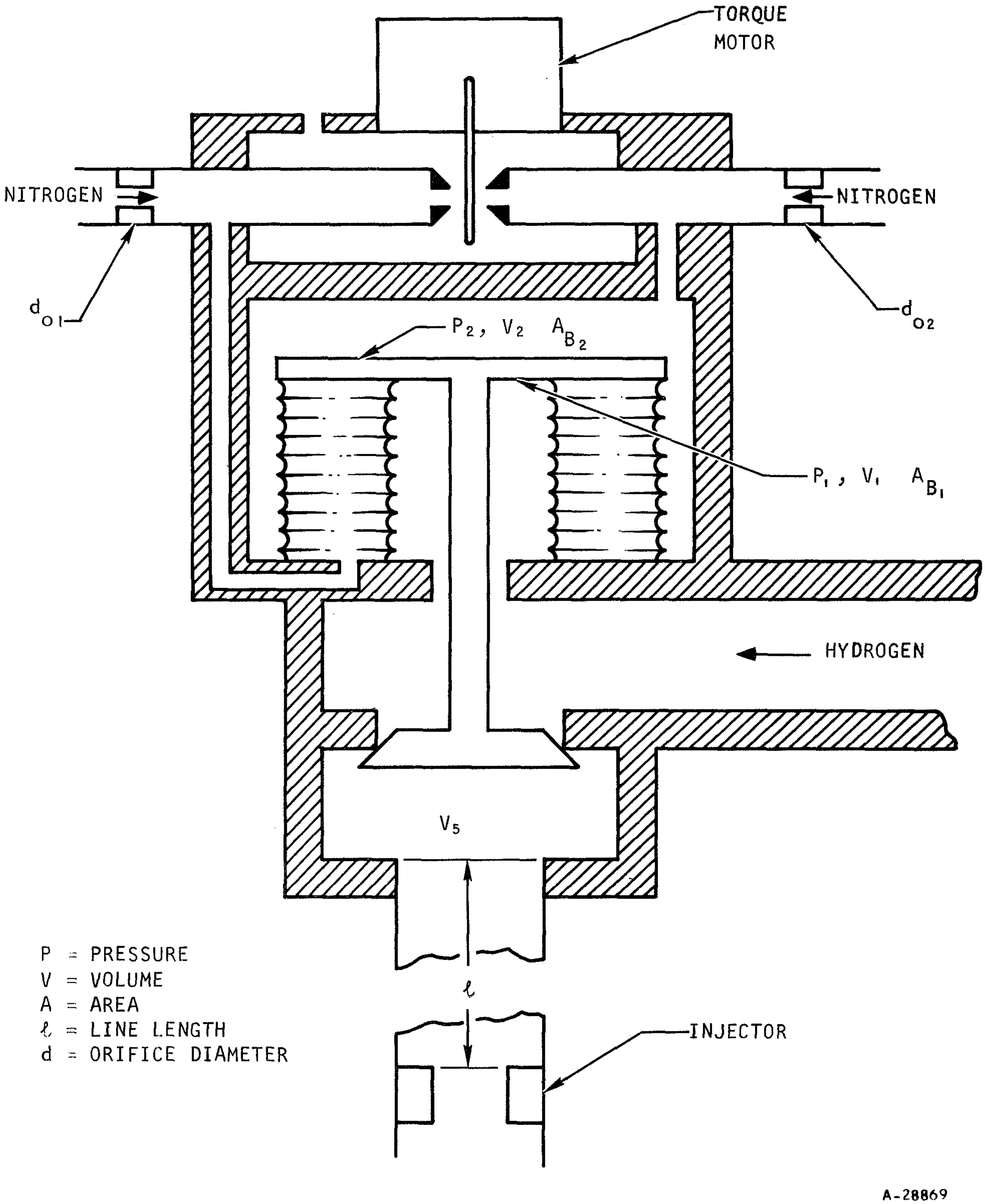

Figure A-1. Schematic Diagram, Hydrogen Flow Control Valve FCV-2

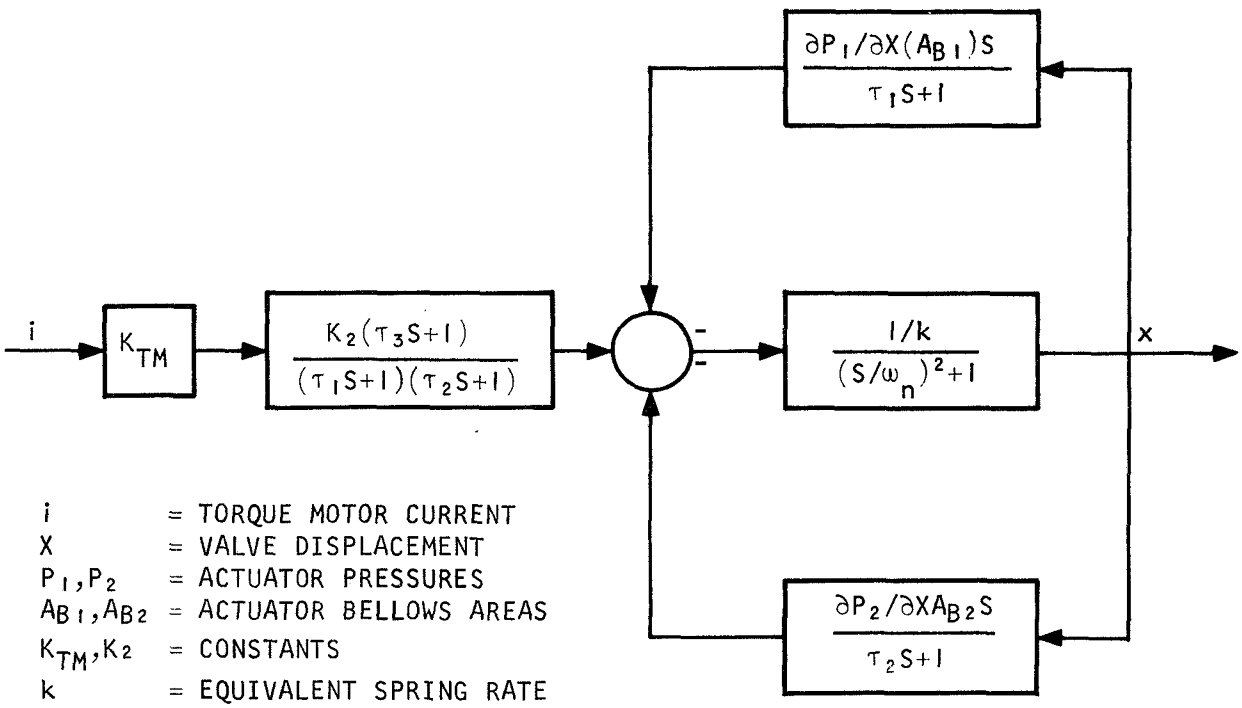

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Figure A-2. Block Diagram, Hydrogen Flow Control Valve FCV-2

TABLE A-1

VALVE FULL OPEN - NO DOWNSTREAM DYNAMIC EFFECTS

d_{o1}, (d_{o2}) in.	τ'_3, sec	τ'_1, sec	τ'_2, sec	ξ	ω'_n, rad per sec	K_2/k
0.011	0.7350	0.4250	15.80	6.22×10^{-4}	862	2230
0.020	0.2340	0.1350	5.05	1.95×10^{-3}	862	1230
0.040	0.0586	0.0337	1.26	7.8×10^{-3}	860	614
0.060	0.0260	0.0150	0.56	1.74×10^{-2}	859	410

Frequency response asymptotes of equation (1) are plotted in Figure A-3. This figure clearly illustrates the frequency response improvement as the inbleed orifice diameter d_{o1} is increased.

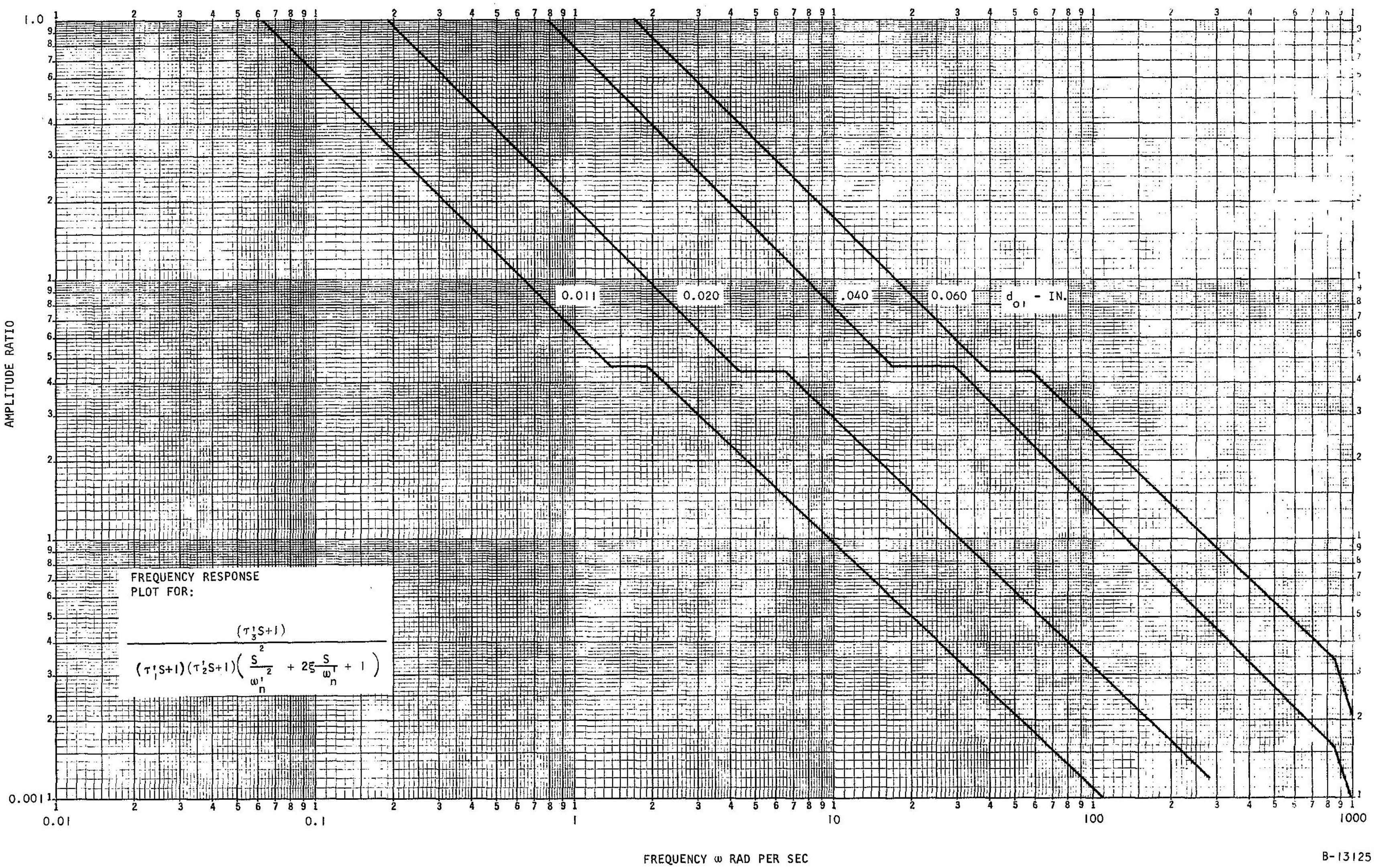

Figure A-3. Valve Frequency Response for Various d_{o_I} Values